



Bottoming cycles for electric energy generation: Parametric investigation of available and innovative solutions for the exploitation of low and medium temperature heat sources

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ABSTRACT

Many industrial processes and conventional fossil fuel energy production systems used in small-medium industries, such as internal combustion engines and gas turbines, provide low or medium temperature (i.e., 200–500 °C) heat fluxes as a by-product, which are typically wasted in the environment. The possibility of exploiting this wasted heat, converting it into electric energy by means of different energy systems, is investigated in this article, by extending the usual range of operation of existing technologies or introducing novel concepts. In particular, among the small size bottoming cycle technologies, the identified solutions which could allow to improve the energy saving performance of an existing plant by generating a certain amount of electric energy are: the Organic Rankine Cycle, the Stirling engine and the Inverted Brayton Cycle; this last is an original thermodynamic concept included in the performed comparative analysis.

Moreover, this paper provides a parametric investigation of the thermodynamic performance of the different systems; in particular, for the Inverted Brayton Cycle, the effects of the heat source characteristics and of the cycle design parameters on the achievable efficiency and specific power are shown. Furthermore, a comparison with other existing energy recovery solutions is performed, in order to assess the market potential. The analysis shows that the highest electric efficiency values, more than 20% with reference to the input heat content, are obtained with the Organic Rankine Cycle, while not negligible values of efficiency (up to 10%) are achievable with the Inverted Brayton Cycle, if the available temperature is higher than 400 °C.

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1. Introduction

In several applications of many industrial sectors the seek for low-cost electric energy generation and the demand for increasing values of the fuel conversion efficiency are emerging requirements, while in most of the cases the heat demanded by process is not a problem as it can be easily covered with high thermal efficiency boilers. Moreover, in some cases low or medium temperature exhaust heat fluxes may be present in the industrial plant, when fossil/renewable fuel engines or gas turbines are used or when the boilers output heat is not fully exploited. For example, an internal combustion engine can provide exhaust gases at temperature values typically of 300–450 °C, a gas turbine is characterized by exhaust temperature of 400–550 °C and micro gas turbines can give 250–350 °C; other industrial heat fluxes, e.g. exhaust from ceramic desiccant ovens, concrete kiln gas, leather or food industry dis-

charge heat, can provide similar temperature values ranging from 200 °C to 500 °C, depending on the process operation.

The available low or medium temperature heat could be profitably exploited by means of a thermodynamic cycle, conceived as bottomer of the heat production process. Typically, in many small-medium size industrial applications, the amount of thermal power discharged by the topping process can be of the order of magnitude of some hundreds of kW, values not compatible with the adoption of superheated water-steam turbine cycles, with complex and multi-level regenerated architecture, typical of large size power plants and granting high efficiency values.

1.1. Brief overview of available technologies for small size heat recovery

Different small size heat recovery systems could be used to exploit the available limited size heat sources, namely Organic Rankine Cycles (ORC) [1] and Micro Rankine Cycles (MRC), Stirling engines [2,3] and also Thermo-Electric (TE) systems [4]; some of these technologies are not yet fully developed (mainly MRC), while

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Nomenclature**Acronyms**

C	compressor
CHP	Combined Heat and Power
IBC	Inverted Brayton Cycle
IC	inter-cooler
LHV	Lower Heating Value
MRC	Micro Rankine Cycle
ORC	Organic Rankine Cycle
REC	recuperator
T	turbine
TE	Thermo-Electric
TI	thermionic
TPV	Thermo Photo Voltaic
Stir	Stirling cycle

Symbols

p_{low}	IBC minimum pressure (bar)
Q_{used}	specific heat at bottoming cycle inlet (kJ/kg)
Q_{avail}	heat available from the source (kJ/kg)
R	gas constant (kJ/kg K)
s	specific entropy (kJ/kg K)
T	temperature (°C)
T_{out}	hot gas outlet temperature (°C)

T_{max}	hot source temperature (°C)
T_{min}	cold source temperature (°C)
T_{vap}	ORC vaporization temperature (°C)
V_{max}	Stirling internal maximum volume (m ³)
V_{min}	Stirling internal minimum volume (m ³)
$X_{H_2O\ in}$	water mass fraction at IBC inlet (–)
$X_{H_2O\ k}$	condensed water mass fraction (–)
$Y_{H_2O\ Sat}$	air saturation water volume fraction (–)
W	specific work (kJ/kg)
W_{Stir_id}	Stirling ideal cycle specific work (kJ/kg)

Greeks

ε	heat recovery effectiveness (–)
η	bottoming cycle efficiency (–)
η_{el}	electric auxiliaries efficiency (–)
η_{gas}	real gas efficiency (–)
η_i	indicator diagram efficiency (–)
η_o	organic efficiency (–)
η_p	polytropic efficiency (–)
η_{rec}	total heat recovery efficiency (–)
η_{rrc}	reversible recuperation cycle efficiency (–)
η_{Stir_id}	Stirling ideal cycle efficiency (–)
η_{II}	irreversibility recuperation cycle efficiency (–)

other are commercially available (ORC, Stirling, TE) but a limited number of producers and of machines could be found (see Table 1 reporting a summary of representative producers and size units for the different considered technologies).

Both the ORC and MRC systems perform a closed thermodynamic cycle similar to the reference Rankine, based on the sequence of: (i) evaporation, exploiting the heat provided by the hot source; (ii) expansion, producing output power; (iii) condensation, discharging unused residual heat; and (iv) fluid pressure augmentation, with pump.

Moreover, the ORC are typically recuperated cycles, to increase the heat recovery and the efficiency (Fig. 1a shows a recuperated ORC layout); refrigerant fluids, hydrocarbon or other particular fluids (siloxanes, i.e. high molecular weight polymerized organic compounds including also silicon and oxygen) are used in the ORC, allowing for dry expansion [5], thus avoiding the problems of turbine operation with a two-phase flow (see a simplified T – s diagram in Fig. 1b, for a dry fluid).

The MRCs, currently under development for output sizes of few kW and oriented to domestic applications, are typically not recuperated, some models are water based (e.g., Otag, Cogen Microsystems), with very simple volumetric expanders and the input heat is provided by a flame, producing a hot gas stream (in most cases inlet temperature can be estimated in the range of 700–800 °C) facing directly the evaporator component, where the steam evaporation

temperature is lower than the hot source temperature and can vary in a wide range, depending on the system design.

The Stirling machine (which can adopt different complicated kinematics structures, [2,3]) is basically an external combustion engine, with an internal fluid (air, helium or other) performing a thermodynamic cycle with values of efficiency which should ideally reach the reference Carnot cycle value; nevertheless, due to the complex architecture of the engine and the strong irreversible processes occurring during the system operation, the actual values of efficiency granted by the existing machines are always lower than 30%; moreover, the current realizations of Stirling engines are operated with an external combustion system, providing the input heat at relatively high temperature (typically 650–800 °C).

The TE systems, based on the semiconductor technology, accomplish the direct conversion of heat into electric power without thermodynamic transformations and without moving parts (the TE exploits the Seebeck–Peltier effect, occurring also under limited temperature differences, down to less than 100 °C, and with hot side temperature ranging between 200 and 400 °C, values compatible with the temperature range in study). Due to its operating principle, the TE system cannot be considered as a bottoming thermodynamic cycle; nevertheless, TE is a possible low temperature heat recovery technology and thus it is included in the present study only for comparison purpose.

Table 1
Small size heat recovery technology producers.

Technology	Producer	Electric power size (kW)	Technology	Producer	Electric power size (kW)
ORC	Turboden (ITA)	200–2000	Stirling	Bioenergy STM (USA)	38, 43
	Infinity turbine (USA)	30–500		WhisperTech Ltd. (NZ)	1
	UTC power (USA)	280		Solo Stirling GmbH (CH)	2–9
	Ormat (USA)	250–2000		Stirling Denmark (DK)	33
	Ingeco (ITA)	100		Microgen (USA)	1
MRC	Energetix group (UK)	2.5	TE	Infinia Corp (USA)	1
	Otag (GER)	2.1		Disenco (GER)	3
	Ormat (USA)	0.2–4.5		Hi-Z technology Inc (USA)	2×10^{-3} – 2×10^{-2}
	Cogen microsystems (AUS)	2.5, 10		Global thermoelectric (USA)	0.02–0.55
				Kryotherm (RUS)	10^{-3} – 10^{-2}

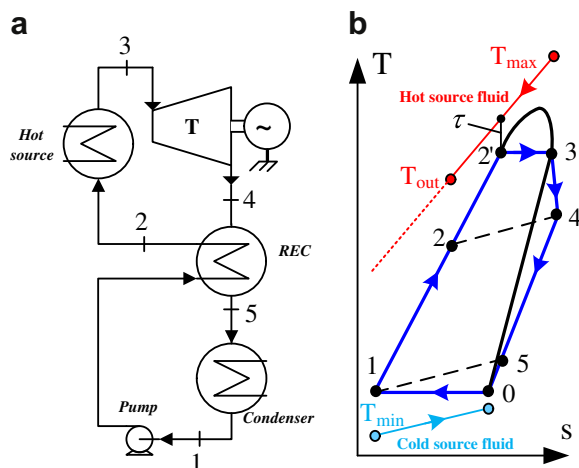


Fig. 1. Recuperated ORC layout (a) and temperature–entropy diagram (b).

1.2. Current limitations of available technologies

As shown in Fig. 2a, the above mentioned technologies are currently covering limited fields of electric power size and each system is designed for a restricted range of temperature of the heat source. In particular, in the range of temperature in study (200–500 °C), the only available solutions are the ORC and the TE technologies, while the Stirling and some of the existing MRC solutions are not designed for the selected range of temperature. In addition, the TE solution is available only for very limited power size (research is going on to increase the power density of this technology [6], but presently the TE commercial systems output power is limited to values equal to few watts), while the ORC solution is available for electric power size normally larger than 100 kW.

Moreover, it is important to notice that, for most of the considered technologies, the output power size sensibly affects the system efficiency, as shown in Fig. 2b reporting the nominal performance of some commercial or under development units; for example, the actual Stirling efficiency drops to values of less than 15%, if the power size reduces to 1 kW and commercial realizations of the ORC show values of about 10%, if the minimum size of 100 kW is reached; typical low size MRC systems provide values below 10% and finally the TE system efficiency is lower than 5%.

Despite these size limitations, the required power size for the heat recovery systems, based on the amount of available heat typically occurring in many industrial applications, can also be of the order of magnitude of 1–100 kW (Fig. 2a).

In this scenario, new technologies or an extension of the temperature/power range for the existing ones should be taken into consideration and the related achievable performance should be assessed. In particular, concerning the ORC and the TE systems, the size range could be extended without modifications in the operating temperature range: the ORC size could be scaled down by reducing the fluid mass flow and the size of the components and the TE system power output could be increased by connecting several modules (the economic feasibility of these strategies, not covered in this paper, should be properly evaluated). On the contrary, concerning the Stirling systems, the engine inlet heat operating temperature should be reduced in order to match the temperature range in study. In this paper, the energetic performance achievable with these operating range extensions is assessed.

1.3. A not yet available heat recovery technology: the Inverted Brayton Cycle (IBC)

Among the not yet developed technologies, the Inverted Brayton Cycle (IBC) is taken into account in this paper as a possible

solution in order to recover heat from a low/medium temperature gas stream available at ambient pressure¹.

In the IBC system [7], as shown in Fig. 3a, the inlet gas stream is expanded in a turbine (T) down to a pressure value below the ambient condition (p_{low}), then it is cooled in a heat-exchanger (IC) and finally recompressed to ambient pressure conditions by a compressor (C) which discharges the gas to the stack. The thermodynamic cycle performed by the gas in the IBC is plotted on a temperature–entropy diagram in Fig. 3b; in the IBC, the cycle maximum temperature is exactly the hot source temperature (T_{max}) due to the upstream process producing the gas, while the cycle minimum temperature is linked with a given delta to the external cooling source temperature (T_{min}), as in the other considered bottoming cycles.

Preliminary analysis on the IBC have been carried out in the past studying the case of IBC as bottomer of a gas turbine (see Tsujikawa et al. [8] and Bianchi et al. [9]) or of a recuperated micro gas turbine [10], or also of a high temperature fuel cell [11]. In these cases, as described in [2,5], the IBC can be also designed for CHP operation, providing thermal energy at very low temperature level (e.g., 60–90 °C) by means of the IC heat-exchanger. In these previous investigations the topping discharged gas temperature was typically of about 500 °C or larger. The range of lower temperature values, not considered in previous analyses, is investigated in this paper.

2. Thermodynamic analysis of available and new heat recovery solutions

In order to assess the potential of the identified small scale bottoming solutions in the field of low/medium temperature, a numerical parametric investigation has been performed. In particular, the effect of the available heat temperature on the recovery performance has been considered. The thermodynamic analysis has been carried out by means of in-house developed routines and with a commercial software (GateCycle™ [14]) by simulating the energy systems with a lumped approach: the main components, namely the heat-exchangers, the expanders and the compressors, have been modelled; the thermodynamic properties of the inlet and outlet streams and the system output work have been calculated on the basis of the component design input data and by applying the mass and energy balance equations to each component; detailed properties of the fluids have been taken into account, considering the specific heats of the gases dependent on the actual temperature and composition and including lookup tables for the organic fluids thermodynamic properties. The thermodynamic properties of the ORC working fluids in the region of liquid, saturated and superheated steam have been calculated by means of the software FluidProp [15].

For the ORC, Stirling and IBC systems, which are based on a thermodynamic cycle, the main key design parameters have been identified and a model of the thermodynamic behaviour of these different systems has been carried out. In particular, the efficiency of the bottoming cycles (η) has been evaluated as follows:

$$\eta = \frac{W}{Q_{used}}$$

where W is the net specific output power and Q_{used} is the input specific heat used by the bottoming cycle. Another performance index useful to compare the analysed heat recuperation systems is the total heat recovery efficiency (η_{rec}), defined as follows:

¹ Other heat recovery technologies now under development like Thermo Photo Voltaic (TPV) and Thermionic (TI) systems are not considered in this study, because their operating range of temperature (typically above 1000 °C [12,13]) is out of the target range of values.

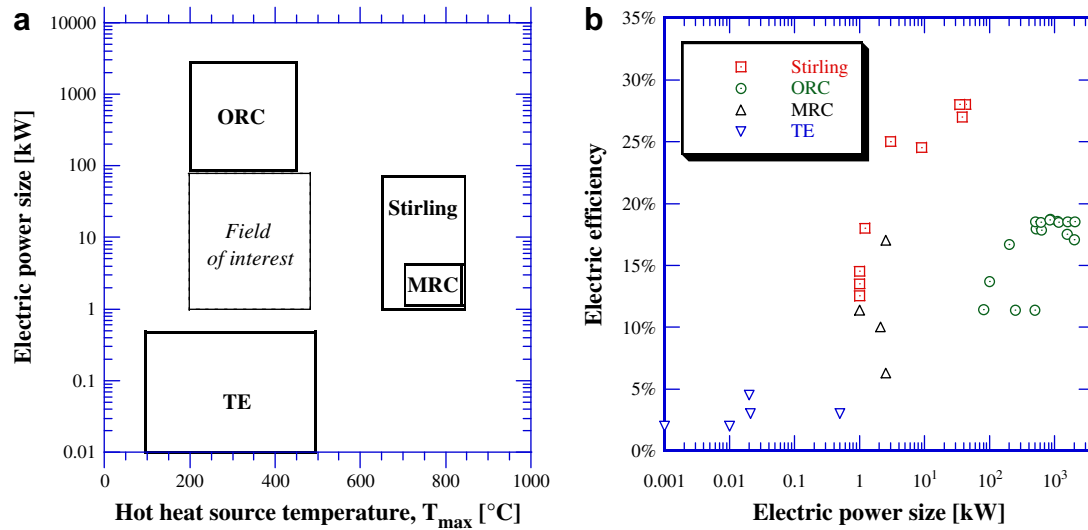


Fig. 2. Range for the hot heat source temperature and for the power size of the available heat recovery technologies (a); efficiency values of some commercial units (b).

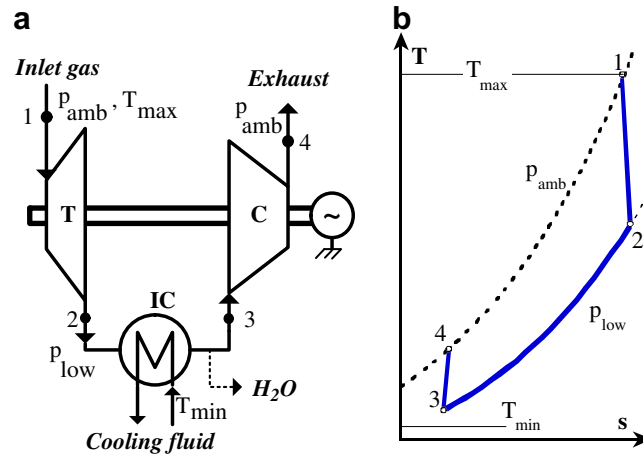


Fig. 3. The IBC layout (a) and the corresponding temperature–entropy diagram (b).

$$\eta_{rec} = \frac{W}{Q_{avail}} = \frac{W}{Q_{used}} \cdot \frac{Q_{used}}{Q_{avail}} = \eta \cdot \varepsilon$$

where ε is the heat recovery effectiveness, equal to the ratio between the heat used in the bottoming cycle and the heat available (Q_{avail}) in the hot stream (gas or other). This index takes into account the fact that, despite the heat recovery in the bottoming cycle, a fraction of the available heat is always wasted, due to the thermodynamic structure of the cycle. The final unused heat is discharged to the environment at a temperature level T_{out} , which is typically higher than T_{min} . In the following sections, the bottoming efficiency is examined for each of the cycles in study and eventually a comparison is carried out also in terms of total heat recovery efficiency.

As a common hypothesis for all the considered systems, the value of the cold source minimum temperature T_{min} has been assumed equal to 15 °C; the effect on the system performance of the alternator and of all the other required electric auxiliaries has been considered in the carried out calculations, by assuming a fixed efficiency value due to the electric auxiliaries equal to 90%.

2.1. ORC modelling

The parametric thermodynamic analysis of the ORC plants has been carried out with the following boundary conditions:

- the investigation has been limited to the case of sub-critical cycles, with saturated steam at the evaporator outlet (for small scale low temperature applications advanced thermodynamic configurations with supercritical or superheated steam could be questionable);
- a recuperated cycle arrangement has been considered, with recuperator effectiveness equal to 80% and recuperator pressure loss equal to 3% of condensing pressure; moreover, the simpler not recuperated case has been evaluated for comparison purpose, to assess the impact of this layout complication on the ORC performance;
- the temperature difference between the inlet hot source temperature (T_{max}) and the ORC saturated steam temperature (T_{vap}) has been fixed equal to 50 °C, in line with typical ORC applications (this fixed difference allows to perform a parametric investigation of the T_{max} value effect, even if an optimization of $T_{max} - T_{vap}$ could be performed for each specific case), while the condensation temperature has been fixed equal to 33 °C, sufficiently higher than T_{min} ;
- four different organic fluids allowing a dry expansion and compatible with the hot source temperature, namely R133a, R245fa, iso-butane and benzene, have been considered, in order to take into account the effect of the fluid properties on the ORC performance (for example, due to the chosen fluids the allowed

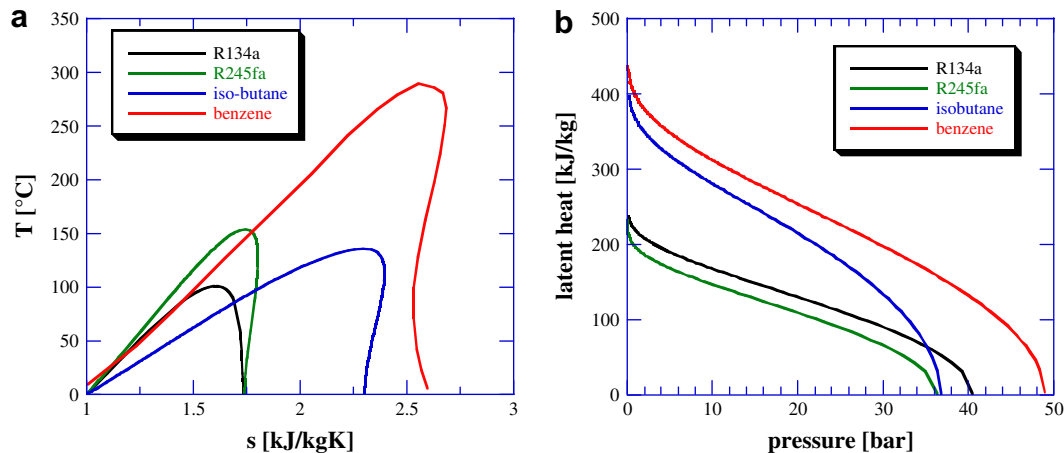


Fig. 4. (a) Saturation lines on a temperature–entropy diagram for the considered organic fluids; (b) latent heats versus pressure.

maximum saturated steam temperature has been varied between 70 °C and 280 °C and the latent heat of condensation at ambient temperature has been varied between 196 and 439 kJ/kg; the fluid saturation lines on the $T-s$ diagram and the corresponding latent heats versus pressure are reported in Fig. 4 and some key thermodynamic properties are collected in Table 2. It should be mentioned that advanced fluid compositions [5], especially developed to improve the thermodynamic properties, have not been taken into account in this study.

- ORC expander isentropic efficiency has been chosen equal to 75%; in other investigations Mago et al. [16] assumed values of 80%, in the framework a parametric analysis of the ORC; Micheli and Reini [17] calculated values in the range 75–85%; the ORC producer *Turboden* declares values up to 90% [18] (efficiency data of some actually built turbomachines for ORC applications are also reported in [19]), but for small size applications a lower range of values should be considered [20];
- mechanical efficiency has been fixed equal to 97%.

The calculated values of the ORC bottoming cycle efficiency (η_{ORC}) and of the specific work per unit of evolving fluid mass flow (W_{ORC}) are given in Fig. 5; each point (η_{ORC} , W_{ORC}) corresponds to a fixed input value of T_{vap} : it can be observed that for all the fluids, increasing the value of T_{vap} up to conditions near the critical temperature, both η_{ORC} and W_{ORC} rise; the maximum values of η_{ORC} and W_{ORC} are dependent on the chosen fluid, with best values $\eta_{ORC} = 23.5\%$ and $W_{ORC} = 140$ kJ/kg obtained with benzene; sub-critical ORCs for higher performances should be designed using fluids with higher critical temperatures and larger latent heats; the choice of the actual fluid is obviously linked to the heat source temperature and constrained by problems of degradation, ozone depletion, global warming impact and safety issues; for example, an organic fluid like R245fa can be less performing in terms of

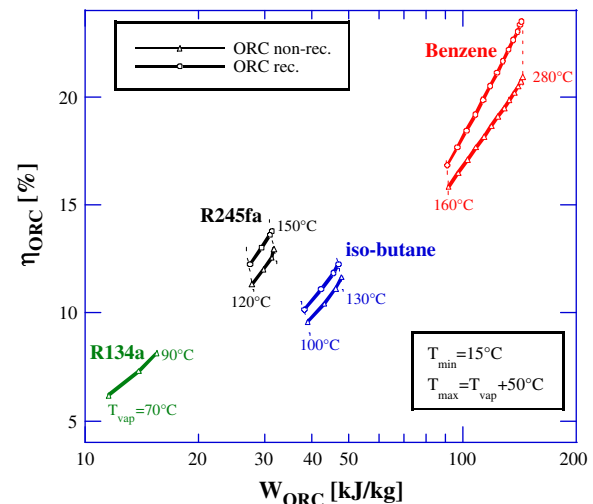


Fig. 5. Efficiency and specific work of recuperated and non-recuperated ORC simulated cases; the saturated steam temperature (T_{vap}) is indicated in figure.

achievable W_{ORC} but also less dangerous in terms of flammability than a hydrocarbon like benzene.

In the simulated cases, the recuperated cycle arrangement grants gains in efficiency of 1–3% points, while the specific power does not change significantly: the simulations show that the effect of the recuperator pressure loss on the power output is limited. The heat recovery process, instead, is affected by the recuperator, which increases the hot source final temperature. For the case of R134a, only the non-recuperated ORC plot is shown in Fig. 5; with this working fluid the recuperated cycle is not convenient, because the steam at the turbine outlet is at the minimum cycle temperature, due to the shape of the fluid saturated vapour line presenting a negative slope (Fig. 4a).

2.2. Stirling modelling

In order to obtain, independently on the engine particular architecture, a simple trend of efficiency and specific power of the Stirling machine versus T_{max} , to evaluate the possibility to recover heat from a topping process, a straightforward model of the engine thermodynamic performance has been used. In particular, the parametric investigation has been performed under the following assumption:

Table 2
Fluids considered for ORC and properties.

Fluid	R134a	R245fa	iso-butane	benzene
Molecular weight (g/mol)	102	134	58	78
Liquid specific volume at 15 °C (dm ³ /kg)	0.822	0.717	1.77	32.9
Latent heat at 1 bar (kJ/kg)	218	196	341	438.7
Critical temperature (°C)	101.1	154.1	135.9	289.0
Critical pressure (bar)	40.6	36.4	36.8	49.0

- a representative Stirling engine has been assumed, operating with $T_{max} = 800$ °C and granting an efficiency value equal to 28% (a value in line with the current models proposed by different producers, see Fig. 1b), when it is fed by the combustion of an external fuel;
- the Stirling performance at lower T_{max} values than the representative engine case has been calculated by means of the following expressions for the efficiency and specific work per unit of internal fluid mass:

$$\eta_{Stir} = \eta_{Stir_id} \cdot \eta_i \cdot \eta_{gas} \cdot \eta_o \cdot \eta_{el}$$

$$W_{Stir} = W_{Stir_id} \cdot \eta_i \cdot \eta_{gas} \cdot \eta_o \cdot \eta_{el}$$

where η_{Stir_id} is the Stirling ideal cycle efficiency (equal to the Carnot expression), η_i is the indicator diagram efficiency, η_{gas} is the real gas efficiency, η_o is the organic efficiency and η_{el} the electric auxiliaries efficiency. The quantity W_{Stir_id} is the Stirling ideal cycle specific work and it can be written as:

$$W_{Stir_id} = R \cdot (T_{max} - T_{min}) \cdot \ln(V_{max}/V_{min})$$

where R is the gas constant and V_{max} and V_{min} are respectively the maximum and minimum engine internal volume;

- both η_o and η_i have been varied in a range of values suggested by Naso [3] typical for Stirling engines: the maximum values have been associated to $T_{max} = 800$ °C and the minimum values to $T_{max} = 200$ °C; the assumed trend of η_o and η_i versus T_{max} is shown in Fig. 6;
- the η_{gas} and η_{el} values have been kept constant equal to 95% and 90% respectively.

The resulting trend of the Stirling cycle efficiency versus T_{max} provided by the considered model is plotted in Fig. 6. The calculated efficiency value is lower than 20%, if the available T_{max} value is lower than 600 °C, and it drops down to less than 10% when T_{max} is reduced below 300 °C. This strong efficiency decrease is mainly due to the unavoidable ideal cycle efficiency drop and secondly by the trend of η_i , conservatively estimated in this study (the η_i penalization could be less dramatic in specific cases of high level technology machine design).

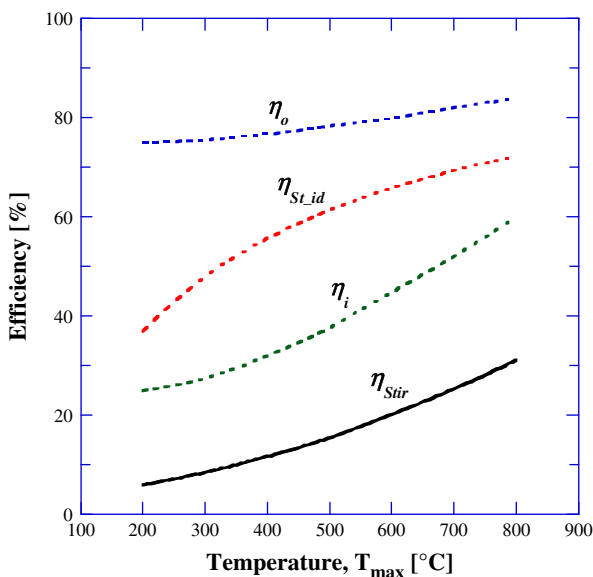


Fig. 6. Stirling efficiency model versus inlet temperature.

Table 3

IBC assumptions.

Parameter	Values
IBC gas inlet pressure	1.013 bar
IBC gas minimum temperature	25 °C
IC gas side pressure drop	1%
Mechanical efficiency	98%

2.3. IBC with low/medium temperature heat source

The IBC potential to exploit a 200–500 °C gas flow is investigated here, calculating the IBC thermodynamic performance in terms of efficiency and specific output work; in the framework of a parametric analysis, the effect of the following design parameters have been considered: (i) the inlet hot gas temperature (T_{max}), the IBC minimum pressure (p_{low}) and the polytropic efficiencies of the IBC turbine and compressor (η_{pT} , η_{pC}). Some fixed parameters taken into account in simulating the IBC system are collected in Table 3.

The calculated values of the IBC net specific work (W_{IBC}) and efficiency (η_{IBC}), obtained for different values of T_{max} , are plotted versus p_{low} in Fig. 7, for constant $T_{min} = 15$ °C of the cold source and fixed minimum temperature of the internal gas equal to 25 °C. The minimum considered p_{low} value has been fixed to 0.3 bar, corresponding to a pressure ratio of about 3.3, a value which can be still accomplished by a single stage centrifugal compressor or by few stages of an axial compressor. For sake of simplicity, the compressor and turbine polytropic efficiency values have been assumed equal to each other and fixed to a value ($\eta_p = 80\%$) compatible with a medium level technology for the turbomachinery design and also in line with a correlation between the machine size and efficiency proposed by Consonni and Macchi [21].

It can be observed that, for a given T_{max} value, a maximum value of work can be reached: the maximum work is obtained for a p_{low} value which tends to be lower when T_{max} is higher; on the contrary, when T_{max} is very low the specific work can be negative (i.e., the IBC is not feasible). The maximum values of specific work, nearly 30 kJ/kg, can be obtained if the gas enters the IBC at the maximum temperature of 500 °C; the corresponding maximum IBC efficiency is equal to 5%.

The effect of the polytropic efficiency value on the IBC thermodynamic performance has been also investigated; the results of a parametric variation of η_p are presented in Fig. 8, where the maximum achievable specific work and the corresponding maximum efficiency values are plotted versus the IBC inlet gas temperature for different values of polytropic efficiency: for a fixed value of T_{max} , the increase in η_p causes a rise of the IBC specific work and efficiency; values close to 70 kJ/kg could be obtained if $\eta_p = 90\%$.

The results of the IBC thermodynamic analysis in Figs. 7 and 8 basically show that the achievable performance are limited and, if the inlet gas temperature is very low (i.e., in the range 200–300 °C), a positive output work could be hardly obtained. In order to increase the cycle thermodynamic performance, the compressor absorbed power should be minimised. A possible strategy consists in reducing the compressor inlet mass flow in comparison with the expander inlet gas mass flow. This aim can be easily accomplished if the IBC inlet gas is rich in water content (for example this could occur in industrial processes where hot dry air is used to exsiccate materials and it is evacuated still hot and wet); in this case, a fraction of the water content can condense at the IC outlet and it can be discharged out of the gas stream (as shown in Fig. 3). In particular, if T_3 is kept largely below 100 °C by means of the inter-cooler, over saturation condition occurs in the gas stream at the IC outlet, leading to water condensation and consequent reduction of the gas mass flow.

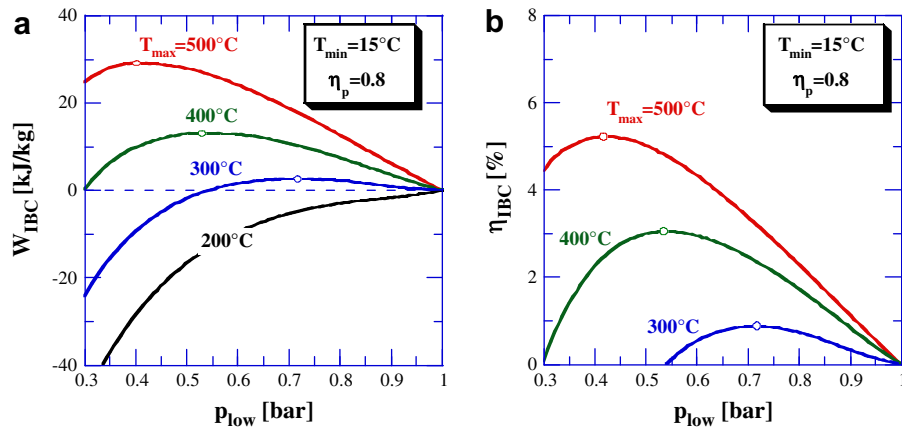


Fig. 7. Specific work (a) and efficiency (b) versus the IBC minimum pressure.

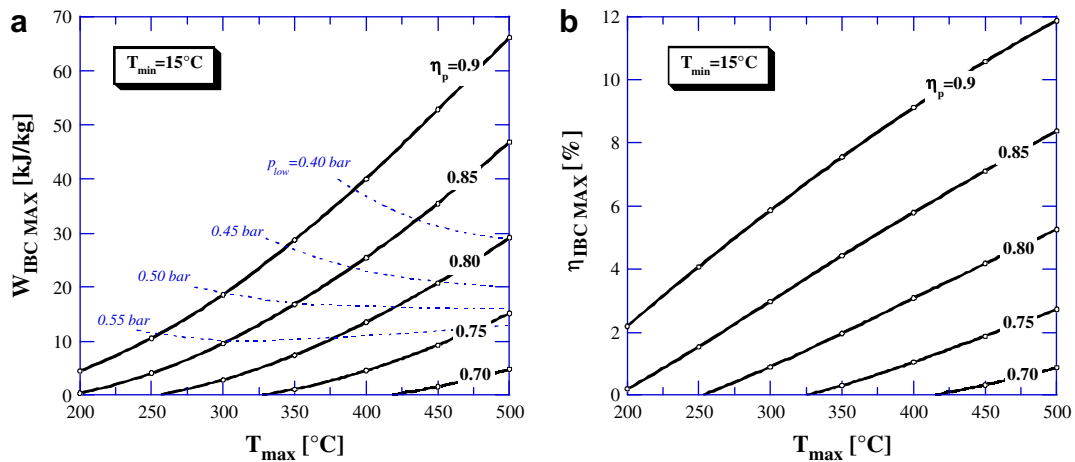


Fig. 8. Effect of polytropic efficiency variation on maximum specific work (a) and maximum efficiency (b).

Fig. 9 shows the air saturation water fraction ($Y_{H_2O Sat}$) as a function of T_3 and of p_{low} ; at $T_3 = 25^\circ\text{C}$ for example, the fraction of dispersed water in the gas is equal to about 3% at ambient pressure, while it becomes equal to about 8% if $p_{low} = 0.4$ bar; in general terms, the lower is p_{low} , the lower the potential reduction of mass flow at the IBC compressor inlet. Moreover, it has been verified that for all the considered values of T_{max} and p_{low} the water condensation can occur only at the IC outlet, because the gas temperature in the upstream sections (in particular at the turbine outlet section) is larger than the condensation temperature.

The actual mass flow reduction at the compressor inlet (due to the surplus in respect to saturation conditions) depends first of all on the water mass fraction at the IBC inlet ($X_{H_2O in}$); in this study, a range of values of $X_{H_2O in} = 0\text{--}30\%$ has been investigated. For a given value of this $X_{H_2O in}$ water content, $X_{H_2O k}$ depends on the T_3 and p_{low} values (which affect the water saturation fraction, according to Fig. 9; the upper limit of $X_{H_2O k}$ is represented by the difference between $X_{H_2O in}$ and the water saturation mass fraction²).

The calculated values of specific work and efficiency are plotted in Fig. 10, versus the mass fraction of condensed and extracted water ($X_{H_2O k}$), for different T_{max} values and for fixed T_{min} and polytropic efficiency; for a given T_{max} , both specific work and efficiency

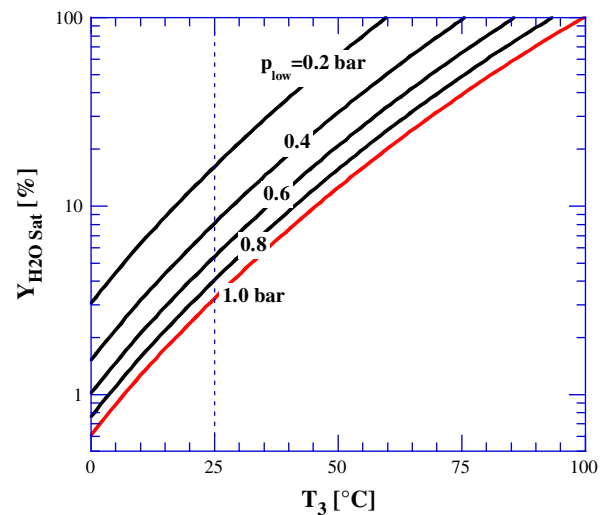


Fig. 9. Water saturation volume fraction versus temperature, for different pressure levels.

increase with the extracted water mass flow rate while the optimum p_{low} value tends to decrease. For the largest considered T_{max} values both $W_{IBC MAX}$ and $\eta_{IBC MAX}$ rise significantly, achieving values respectively equal to 70 kJ/kg and 10% when $X_{H_2O in} = 30\%$.

² The desired amount of water extraction can be realised with a proper design of the heat exchanger internal surfaces and with limited air velocity.

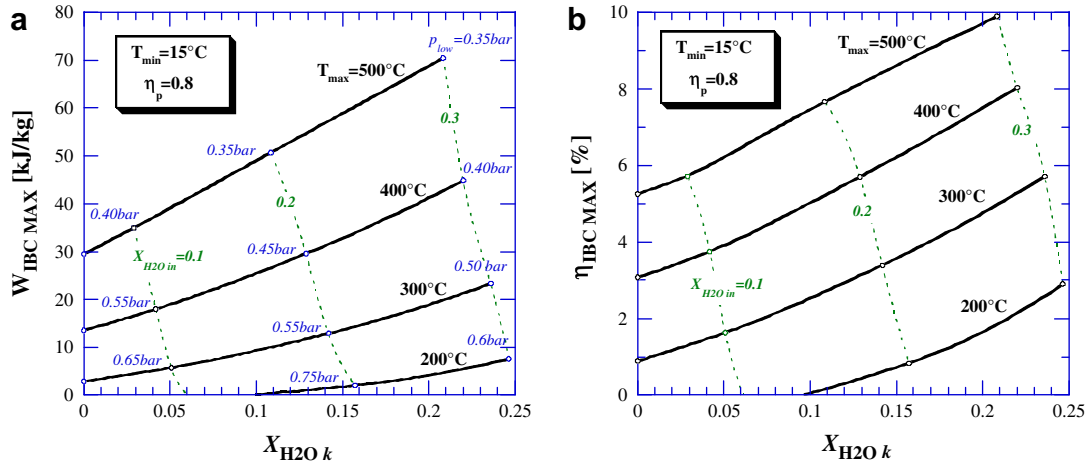


Fig. 10. Maximum specific work (a) and maximum efficiency (b) for different amount of condensable water at the IC outlet.

3. Comparison of the bottoming cycles

A comparison of the different analysed heat recovery technologies is carried out in this section. The results of efficiency versus T_{max} for the different solutions are shown in Fig. 11: a unique continuous line is shown for the ORC, which interpolates the points corresponding to the efficiency values calculated for the four different considered fluids in the range of T_{max} equal to 130–330 °C (compatible with the chosen fluids). The Stirling engine curve corresponds to the assumptions reported in the previous paragraphs.

Two continuous lines are plotted for the IBC system without water condensation, corresponding to different polytropic efficiency values, namely 80% (middle level turbomachinery design) and 90% (advanced blading design). In particular, the shown values correspond to a IBC design with p_{low} of maximum cycle efficiency for given T_{max} and η_p (as shown before, for a T_{max} lower than 400–500 °C, the optimum p_{low} is typically larger than 0.3 bar). Also, two lines for the case of IBC with water condensation (IBC + H₂O cond.) are provided in Fig. 11; the efficiency values on these curves have been calculated assuming $X_{H_2O in} = 20\%$.

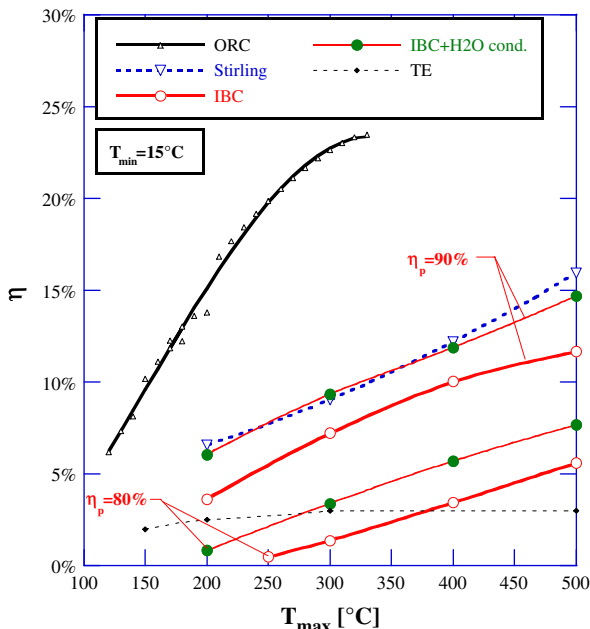


Fig. 11. Bottoming cycle efficiency versus T_{max} .

Finally, the trend of efficiency versus T_{max} for the TE systems is shown in Fig. 11 for comparison purpose; this trend has been obtained by interpolating different commercial unit data.

In the investigated range of temperature the ORC is the solution granting the highest bottoming cycle efficiency values (up to about 23% when T_{max} is 300 °C or higher and close to 15% when T_{max} reduces to 200 °C); the Stirling engine, instead, even if at high temperature is the most efficient system, undergoes a dramatic efficiency drop if T_{max} decreases.

The IBC can provide bottoming cycle efficiency values up to 12% in case of larger T_{max} and η_p values, but if the inlet temperature is low the employment of IBC becomes questionable. Instead, the IBC efficiency increases of 1–2 percentage points if the water condensing arrangement is applied.

The results of total heat recovery efficiency are reported in Fig. 12 for all the investigated systems. The values of η_{rec} are calculated, in case of ORC systems, assuming a fixed pinch point (τ , shown in Fig. 1b) equal to 10 °C at the vaporizer inlet. It can be seen that in the ORC cases the T_{max} value has a strong impact on the total heat recovery efficiency, for a fixed organic fluid; indeed, the higher is T_{max} , the larger the value of η_{rec} , because η rises and at the same time T_{out} decreases (because $T_{max} - T_{vap}$ and τ are fixed). The highest values of η_{rec} , equal to about 21%, are obtained with benzene and with $T_{max} = 330$ °C. The values of η_{rec} reported in Fig. 12 for the Stirling system are calculated assuming a constant heat recovery effectiveness value equal to 0.9.

As a general result, the efficiency values calculated for all the systems tend to reduce when T_{max} is decreased, but also the maximum obtainable work with an ideal reversible process would decrease if T_{max} is reduced. Therefore, in order to quantify and compare the losses due to irreversible process occurring in the various bottoming cycles, independently on the T_{max} level, a different performance indicator, named here irreversibility recuperation cycle efficiency (η_{II}), has been also evaluated. The value of η_{II} has been calculated according to the following equation:

$$\eta_{II} = \frac{\eta}{\eta_{rrc}} = \frac{\eta}{\left(1 - \frac{T_{min}}{T_{max} - T_{min}} \cdot \ln \frac{T_{max}}{T_{min}}\right)}$$

where the denominator η_{rrc} is the efficiency of an ideal reversible recuperation cycle (made up of a isobaric heat absorption, an isentropic expansion and an isothermal compression, see Figs. 13 and 14) in which, during the process of heat exchange with the cycle evolving fluid at constant pressure, the hot source temperature decreases from T_{max} to T_{min} . Thus, η_{II} represents a measure of the distance of the actual thermodynamic cycle from the ideal cycle in

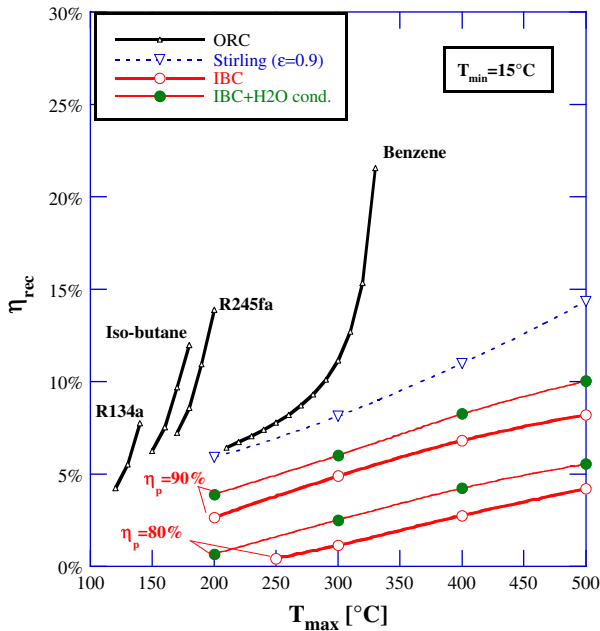


Fig. 12. Total heat recovery efficiency versus T_{max} .

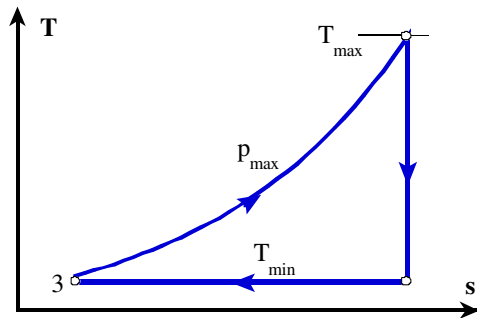


Fig. 13. T - s diagram of the ideal reversible recuperation cycle to be considered when the hot source temperature decreases.

which heat is exchanged with a finite, variable temperature, heat source.

In particular, the obtained results of η_{II} versus T_{max} , plotted in Fig. 15, show that the ORC solution provides the highest η_{II} values, with remarkably high values close to 80%³. The Stirling engine values are in the range of 30–40%, while the IBC values can be up to 30% in dry gas conditions, or even close to 35% in the water condensing arrangement, if the design is advanced ($\eta_p = 90\%$) and $T_{max} = 500^\circ\text{C}$. The larger values of ORC systems is due to the fact that a Rankine cycle tends to resemble an ideal Carnot cycle more than the other examined solutions. For IBC, the lower value of η_p , corresponding to higher irreversibility in the compression and expansion process, leads to a strong reduction in efficiency; water condensation, instead, reduces the irreversibility of the final compression process. The TE systems η_{II} values, equal to about 10% at the lowest T_{max} values, are significantly lower than the corresponding values for ORC and Stirling.

Fig. 16 provides a comparison of the different thermodynamic bottoming cycles in study in terms of specific work, which gives

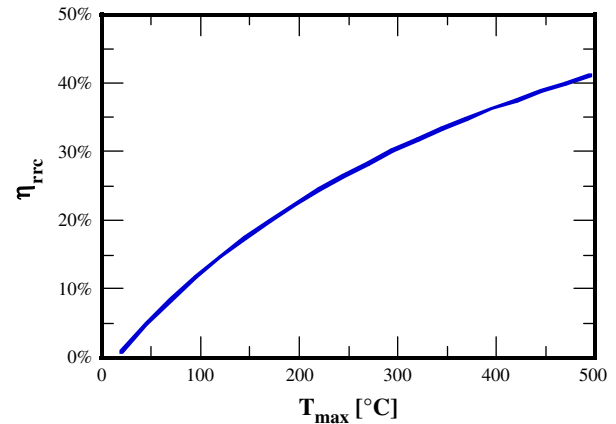


Fig. 14. Efficiency of the ideal reversible recuperation cycle.

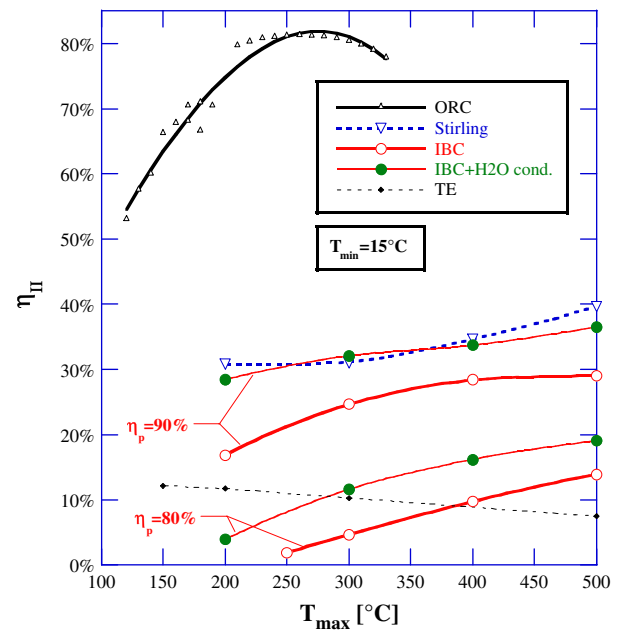


Fig. 15. Irreversibility recuperation cycle efficiency of the considered bottoming cycles versus T_{max} .

information on the system size. In particular, the specific work is plotted versus T_{max} for the ORC, the IBC and the Stirling systems. The results show that W strongly depends on the operating fluid for all the considered bottoming cycles; in the case of ORC, four different curves are plotted for the four analysed organic fluids; for the Stirling engine air and helium have been considered, while for the IBC only the case of air has been investigated. In the range of $T_{max} = 200$ – 350°C , if the fluid is accurately designed, the ORC specific work can be larger than the values provided by both Stirling and IBC. The Stirling specific work, which is as large as 100–400 kJ/kg when the engine is designed to operate at $T_{max} = 800^\circ\text{C}$, decreases down to values of 10–40 kJ/kg when $T_{max} = 200^\circ\text{C}$. The specific work values for the IBC, calculated in the same design points of Fig. 11, shows the largest reduction when T_{max} decreases.

Finally, it should be mentioned that, in order to perform a full comparison of the technologies, the heat recovery efficiency and the specific energy production are not the only aspects to be evaluated; in particular, the marginal investment cost is another key issue. Although a proper economic analysis has not been introduced in this study, it should be evident that the heat recovery systems

³ It should be noted that if the Carnot efficiency is used instead of η_{rrc} the η_{II} values would be lower.

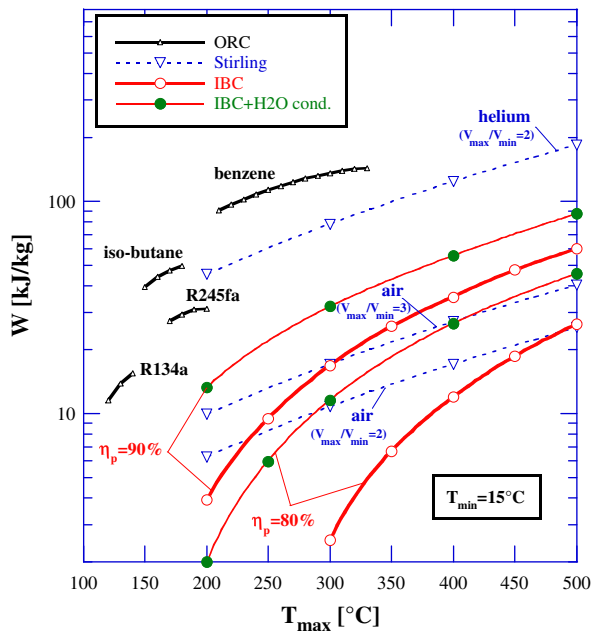


Fig. 16. Specific work of the considered bottoming cycles versus T_{max} .

are almost free of variable costs (no need of additional fuel consumption); therefore, a final driver of the choice among the bottoming solutions is the ratio of the investment cost to the value of the additional produced energy. The IBC system could result convenient if its investment costs are very low, even if it is the solution granting the lowest specific work values in many T_{max} cases.

4. Conclusions

In the framework of the performed thermodynamic investigation, the ORC technology results as the most performing and well proven solution, in order to exploit low/medium temperature heat sources, even if currently it is not yet developed for small scale applications. The selection of adequate organic fluids represents a key factor to maximize the ORC thermodynamic performance which can be significant, as shown in this study, even if a small size (and relatively low efficiency) turbine is used.

The Stirling engine, despite its promising performance in terms of efficiency and specific power when operated with hot thermal source (temperature of the order of 800 °C), shows a drastic performance penalization if connected with lower temperature heat sources (200–500 °C) and does not seem the most promising solution to recover a wasted heat flow supplied by a topping industrial process.

Among the investigated heat recovery strategies, the innovative and not yet developed IBC system is a promising solution but not as performing as the ORC technology, especially in the field of very low temperatures (200–400 °C). If instead heat fluxes are available at temperature values above 350–400 °C, the IBC technology becomes more interesting in terms of achievable efficiency. Moreover, the IBC system allows a cogenerative operation with thermal power production at the IC heat-exchanger, not investigated in this study. The IBC components are well known and available in other applications and do not require advanced technology (in a first prototype developing phase, the possibility to reutilize in a IBC small turbochargers or microturbine components could be assessed); nevertheless, in order to maximize the system performance an optimized design of the IBC turbomachinery would be required.

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